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# Simulation of the thermosyphon free cooling mode in an integrated system of mechanical refrigeration and thermosyphon for data centers

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**Abstract**

Integrated system of mechanical refrigeration and thermosyphon (ISMT) is an ideal solution for energy-saving of data centers. The performance of the thermosyphon mode of an ISMT is very important and has great effect on its annual energy-saving rate. To investigate and optimize the performance of the thermosyphon mode, a distributed-parameter model is built and verified by experimental results. With this model, the performance of different air flow rate and geometric parameters is studied. The results show that the cooling capacity and circulation flow rate both increase with air flow rate and riser diameter, and decrease with tube length. Combined with these performance simulation results, the annual working time of thermosyphon mode is also calculated for a small data center in Beijing. The method to achieve the optimal design is given for different design requirements.


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*Keywords:* thermosyphon; free cooling; data center; simulation

## 1. Introduction

The number of data centers is increasing rapidly in recent years. Electricity consumed by data centers occupied 1.3% of all electricity use for world in 2010 [1]. Data center must be adequately cooled and now most data centers are equipped with computer room air conditioner (CRAC). However, traditional CRAC is not efficient and its consumption takes up approximately 50% of the total consumption of in a data center [2]. Cutting down the energy used by CRAC is an urgent need and several methods have been proposed [3-5]. Among these methods, free cooling based on thermosyphon loop shows great application

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potential for its high efficiency and no disturbance on the indoor environment. As thermosyphon can only work in cool seasons, mechanical refrigeration systems have to be equipped for hot seasons. To avoid two sets of equipment, integrated system of mechanical refrigeration and thermosyphon (ISMT) is an ideal solution and several designs have been proposed [6-8]. Zhang et al. proposed a new type ISMT, which overcomes the short comings of the former design and achieves high energy efficiency [9].

The cooling capacity of thermosyphon mode of an ISMT is an important performance parameter and has significant effect on the energy-saving rate. Experimental studies on the optimization of thermosyphon mode of ISMTs have been conducted by researchers [6-9]. Until now, there is no published literature on simulation model of ISMT. Although the simulation of loop thermosyphon have been studied by many researchers, few of them focused on air cooled ones, which has unique features such as uncertain heat input and small heat flux. Moreover, the former studies did not give an overall consideration of the performance and energy consumption, which is important for application.

In this paper, a simulation model of the thermosyphon loop in the ISMT proposed by Zhang et al. [9] is built and verified. The corresponding performance and energy-saving rate is calculated and compared. This paper offers a new method and meaningful results for optimization of ISMTs.

**Nomenclature**

| | | | |
|---|---|---|---|
| $F$ | filling quantity of working fluid, kg | $Q$ | heat transfer rate, W |
| $H$ | enthalpy, kJ $kg^{-1}$ | *Subscripts* | |
| $m$ | mass flow rate, kg $s^{-1}$ | act | actual |
| $P$ | pressure, Pa | cal | calculated |
| $\Delta p$ | pressure drop, Pa | sup | supposed |

## 2. Simulation model

A simulation model of the thermosyphon mode of an ISMT is built. The ISMT proposed by Zhang et al. [9] is shown in Fig. 1. The system consists of two circulation loops: a refrigeration loop and a thermosyphon loop. A three-fluid heat exchanger (THE) is used to connect the two loops, which has three flow channels for the fluid of the refrigeration loop, the fluid of thermosyphon loop and outdoor air respectively. This system has three working modes. It works in refrigeration mode in hot weather, the fan of the three-fluid heat exchanger stops and the compressor starts to cool the thermosyphon working fluid. Dual mode in mild weather, all the fans and the compressor work to utilize both natural cold source and mechanical refrigeration cold source. Thermosyphon mode in cold weather, only the fans of three-fluid heat exchanger and evaporator work to cool the data center using natural cold air.

The detailed geometric parameters of the evaporator and the three-fluid heat exchanger can be referred to Ref. [9]. The geometric parameters of the connection pipes and the inlet conditions of the thermosyphon loop used for simulation are shown in Table 1. Some values will be changed in the following chapters to study their effect on the performance and the rest ones keep the value in Table 1.

The numerical model is developed with distributed-parameter method. The flow chart of simulation is shown in Fig. 2. The internal flow parameters of the thermosyphon loop are unknown therefore three iterations are needed. The calculated filling quantity is the sum of the fluid mass of all control volumes. The evaporation heat transfer coefficient is calculated according to the correlation proposed by Kandlikar [10]. The correlation proposed by Jaster and Kosky [11] is used to calculate the condensation heat transfer coefficient. The airside heat transfer coefficient is calculated using the equation recommended in Ref.

[12]. The friction loss of two-phase flow is calculated by Lockhart–Martinelli Equation [13].

In order to verify the accuracy of the model, the calculated heat transfer rate and system pressure is compared to our experimental results of thermosyphon mode of the ISMT as shown in Fig. 3. The calculated values agree well with the experimental ones with deviations of ±5%.

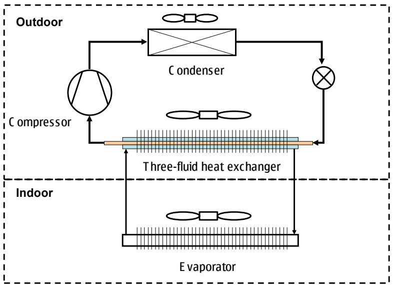


Fig. 1. Integrated system of mechanical refrigeration and thermosyphon

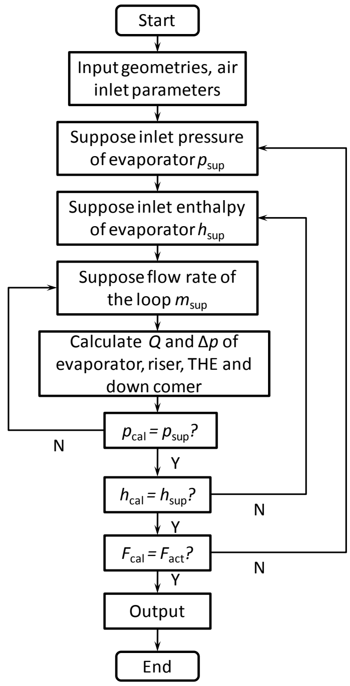


Fig. 2. Flow chart of the simulation

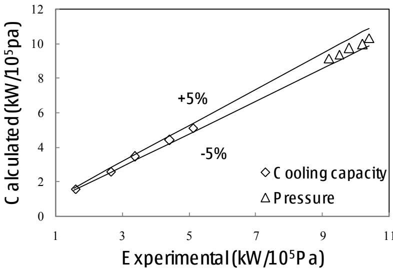


Fig. 3. Comparison of calculated and experimental values of cooling capacity and pressure

Table 1. Geometric parameters and inlet conditions

| Parameter | Value | Parameters | Value |
|---|---|---|---|
| Riser length/outside diameter (m) | 1.0/0.019 | Air flow rate of evaporator ($m^3$ $s^{-1}$) | 0.75 |
| Down comer length/outside diameter (m) | 1.76/0.016 | Outdoor temperature (°C) | 12-22 |
| Height difference (m) | 1.0 | Outdoor relative humidity | 50% |
| Indoor temperature (°C) | 27 | Air flow rate of THE ($m^3$ $s^{-1}$) | 1.114 |
| Indoor relative humidity | 50% | | |

## 3. Results and discussions

### 3.1. Performance analysis

Air flow rate greatly affects the airside heat transfer coefficient and the fluid flow inside the loop. The cooling capacity and circulation flow rate as a function of the indoor air flow rate when the outdoor air flow rate is fixed is shown in Fig. 4, as a function of the outdoor air flow rate when the indoor air flow rate is fixed is shown in Fig. 5. The indoor and outdoor temperature differences are 5 $^{o}$C, 10 $^{o}$C and 15 $^{o}$C. It can be seen that the cooling capacity increases with increasing air flow rate and temperature difference. The increasing air flow rate and temperature difference both promote the fluid flow inside the loop. Higher air flow rate means higher energy consumption of fans therefore it should be determined considering both cooling capacity and energy consumption.

The length and diameter of connection tube are also important parameters. The cooling capacity and circulation flow rate as a function of the riser length are shown in Fig. 6. (a) and Fig. 6. (b), respectively. It is worth noting that when the riser length increases, the down comer length increases with the same increment. It can be seen from Fig. 6. (a) that the cooling capacity decreases with the increasing tube length. It is due to the increasing flow resistance when the tube length increases, which reduces the circulation flow rate as seen from Fig. 6. (b). The cooling capacity and circulation flow rate as a function of riser inside diameter are shown in Fig. 7. (a) and Fig. 7. (b), respectively. The cooling capacity slightly increases with increasing riser diameter due to the decreasing flow resistance. The circulation mass flow rate increases with increasing indoor and outdoor temperature difference.

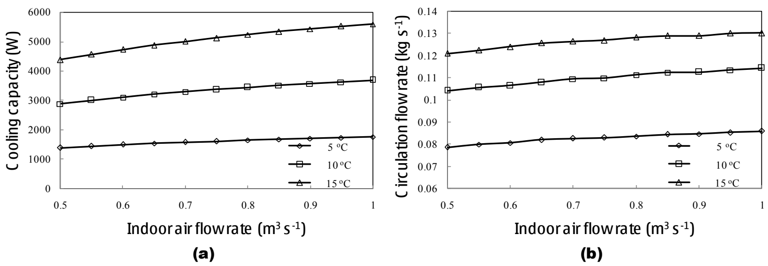


Fig. 4. Cooling capacity and circulation flow rate for different indoor air flow rates

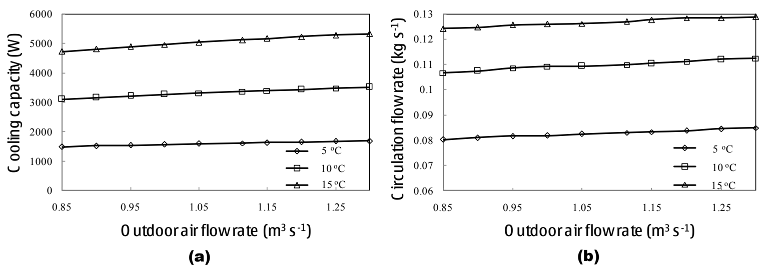


Fig. 5. Cooling capacity and circulation flow rate for different outdoor air flow rates

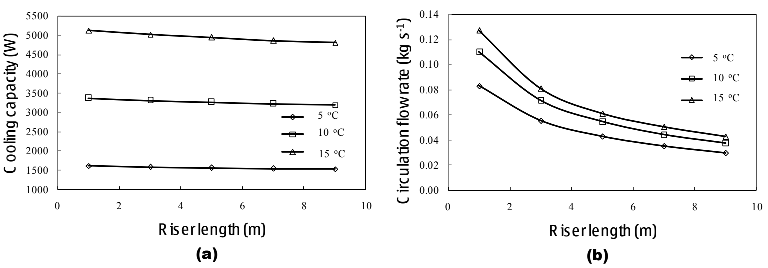


Fig. 6. Cooling capacity/Circulation flow rate for different tube length

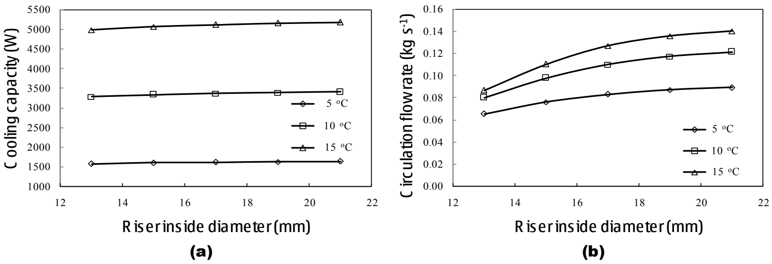


Fig. 7. Cooling capacity/Circulation flow rate for different riser diameter

### 3.2. Annual energy-saving potential

As mentioned above, the determination of air flow rate and structure parameters needs an overall consideration based on energy-saving rate of the ISMT. A small data center in Beijing is chosen to analyze the energy-saving rate. The data of the data center and the climate data of Beijing can be referred to in Ref. [9]. The hourly cooling load of the data center can then be calculated. If the cooling load is lower than the cooling capacity of the thermosyphon mode, the thermosyphon mode works and the compressor stands by. Therefore, the working time of thermosyphon mode reflects the energy-saving rate.

Combined with the performance simulation results in Chapter 3.1, the working time of thermosyphon mode changing with the indoor and outdoor air flow rate is calculated and shown in Fig. 8. When the air flow rate is higher, the cooling capacity of the thermosyphon mode is larger therefore the working time of it is longer. If the optimization objective is to obtain an air flow rate with highest energy-saving rate, it can be achieved by comparing the energy-saving of the working time of thermosyphon mode and the energy consumption of fans. If the optimization objective is to obtain a certain working time, there are more than one groups of air flow rate can be chosen from. For example, if the desired working time of thermosyphon is 5100 h, the indoor and outdoor air flow rate can be set at 0.8 $m^3 s^{-1}$ and 1.114$m^{-3} s^{-1}$, respectively, or 0.75 $m^3 s^{-1}$ and 1.2 $m^3 s^{-1}$, respectively, as marked in Fig. 8. In this method, the optimal air flow rate can be decided by comparing the energy consumption of fans of these groups.

The working time of thermosyphon mode changing with the rise diameter is shown in Fig. 9. The working time increases with increasing riser diameter, while larger diameter means greater investment. The optimal diameter can be obtained by comparing the increase of the investment and the energy-saving of thermosyphon working time.

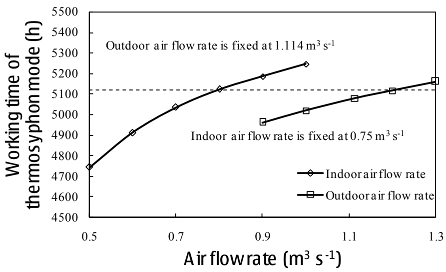


Fig. 8. Working time of thermosyphon mode for different air flow rate

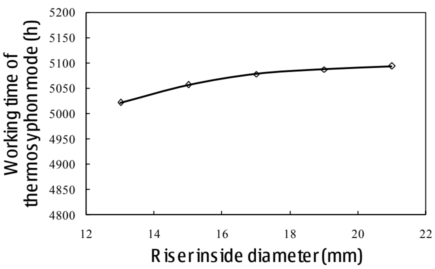


Fig.9. Working time of thermosyphon mode for different air flow rate

## 4. Conclusion

A distributed-parameter model is built for the thermosyphon mode of an ISMT. The calculated values agree with the experimental values with deviations of ±5%. With this model, the performance and energy-saving potential are investigated and the main results are concluded as follows:

(1) The cooling capacity increases with increasing air flow rate and riser diameter and decreases with increasing connection tube length, which is mainly due to the variation of flow resistance and mass flow rate.

(2) The determination of air flow rate and structure parameters needs an overall consideration based on energy-saving rate. The annual working time of the thermosyphon mode of the ISMT of different air flow rate and riser diameter is calculated for a typical data center in Beijing. In this method, the optimal design with the highest energy-saving rate can be determined.

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**Biography**

Shuangquan Shao was born in 1975. He received his Ph.D. in 2005 from Tsinghua University. He is now an Associate Professor of the Technical Institute of Physics and Chemistry, Chinese Academy of Sciences. Prof. Shao's research interests include heat pump, automotive air conditioning, thermal management et al.